Chapter 4: Dry Cooling

Chapter Contents

INTRODUCTION

This chapter addresses the use and performance of dry cooling systems at power plants. Dry cooling systems transfer heat to the atmosphere without the evaporative loss of water. There are two types of dry cooling systems for power plant applications: direct dry cooling and indirect dry cooling. Direct dry cooling systems utilize air to directly condense steam, while indirect dry cooling systems utilize a closed cycle water cooling system to condense steam, and the heated water is then air cooled. Indirect dry cooling generally applies to retrofit situations at existing power plants because a water-cooled condenser would already be in place for a once-through or recirculated cooling system. Therefore, indirect dry cooling systems are not further considered in the Chapter for new sources subject to this regulation.

The most common type of direct dry cooling systems (towers) for new power plants are recirculated cooling systems with mechanical draft towers. Natural draft towers are infrequently used for installations in the United States and were not considered for evaluation in this Chapter.

For dry cooling towers the turbine exhaust steam exits directly to an air-cooled, finned-tube condenser. The arrangement of the finned tubes are most generally of an A-frame pattern to reduce the land area required. However, due to the fact that dry cooling towers do not evaporate water for heat transfer, the towers are quite large in comparison to similarly sized wet cooling towers. Because dry cooling towers rely on sensible heat transfer, a large quantity of air must be forced across the finned tubes by fans to improve heat rejection. The number of fans is therefore larger than would be used in a mechanical draft wet cooling tower.

Hybrid wet-dry cooling towers employ both a wet section and dry section and are used primarily to reduce or eliminate the vapor plumes associated with wet cooling towers. For the most common type of hybrid system, exhaust steam flows through smooth tubes, where it is condensed by a mixture of cascading water and air. The water and air move in a downward direction across the tube bundles and the air is forced upward for discharge to the atmosphere. The falling water is collected and recirculated, similarly to a wet cooling tower. The water usage of a hybrid system is generally one-third to one-half of that for a wet cooling system and the required pumping head is reduced somewhat. In the Agency's opinion, the common hybrid systems do not dramatically reduce water use as compared to wet cooling towers. The comparative cost increases of the hybrid systems to the wet cooling systems do not outweigh water use savings of approximately one-half to two-thirds. Therefore, the discussion of dry cooling towers for the remainder of the chapter focuses on direct dry cooling systems exclusively.

The key feature of dry cooling systems is that no evaporative cooling or release of heat to surface water occurs. As a result, water consumption rates are very low compared to wet cooling systems. Since the unit does not rely in principle on evaporative cooling as does a wet cooling tower, larger volumes of air must be passed through the system compared to the volume of air used in wet cooling towers. As a result, dry cooling towers need larger heat transfer surfaces and, therefore, tend to be larger in size than comparable wet cooling towers. The design and performance of the dry cooling system is based on the ambient dry bulb temperature. The dry bulb temperature is higher than the wet bulb temperature under most circumstances, being equal to the wet bulb temperature only when the relative humidity is at 100%.

The remainder of this chapter is organized as follows:

- Section 4.1 provides a brief overview of the status of dry cooling projects in the United States including discussion of the types of generating facilities, their locations, and factors affecting plant performance.
- Section 4.2 presents an evaluation of the dry cooling technology as a candidate for best technology available to minimize adverse environmental impact.

4.1 DEMONSTRATED DRY COOLING PROJECTS

This section provides a brief overview of the status of dry cooling projects in the United States. The section includes a brief discussion of the types of generating facilities, their locations, and factors affecting plant performance.

Dry cooling has been installed at a variety of power plants utilizing many fuel types. In the United States, dry cooling is most frequently applied at plants in northern climates. Additionally, arid areas with significant water scarcity concerns have also experiencing growth in dry cooling system projects. As demonstrated in Chapter 3, the comparative energy penalty of a dry cooling plant in a hot environment at peak summer conditions can exceed 12 percent, and the benefit of the water use savings must be analyzed with regard to the reduced cooling efficiency.

Table 4-1 presents a compilation of data pertaining to dry cooling systems installed at power plants within the United States and in foreign countries by a U.S. dry cooling system manufacturer from 1968 through the year 2000. The majority of these systems have been installed at combined cycle plants and at alternative fuel plants such as municipal solid waste and waste wood burning facilities. In many cases, systems with similar design dry bulb temperatures have different design exhaust pressure values, reflecting the selection of different dry tower sizes by the facility owners. Use of different relative dry tower sizes for similar facilities reflects the selection of different economic criteria with respect to size, costs, and efficiency.

Table 4-1: Air Cooled Condenser Data for Systems installed by GEA Power Cooling Systems, Inc.

Facility Name	City	State	Country	Size MW	Steam Flow lbs/hr	Turbine Exhaust Pressure In. Hg	Design Temp. °F	Year	Description	Sat. Steam Temp. °F	Temp. Difference °F
Neil Simpson I Sta.	Gillette	WY	USA	20	167,550	4.5	75	1968	Coal	130	55
NP Potter	Braintree	MA	USA	20	190,000	3.5	50	1975	Combine Cycle	120	70
Wyodak Sta.	Gillette	WY	USA	330	1,884,800	6	66	1977	Coal	141	75
Gerber Cogen	Gerber	CA	USA	3.7	52,030	2.03	48	1981	Combined Cycle Cogen	102	54
NAS North Is. Cogen	Coronado	CA	USA	4	65,000	5	70	1984	Combined Cycle Cogen	134	64
NTC Cogen	San Diego	CA	USA	2.6	40,000	5	70	1984	Combined Cycle Cogen	134	64
Chinese Sta.	China Camp	CA	USA	22.4	181,880	6	97	1984	Waste wood	141	44
Duchess Cnty. RRF	Poughkeepsie	NY	USA	7.5	50,340	4	79	1985	WTE	126	47
Sherman Sta.	Sherman Station	ME	USA	20	125,450	2	43	1985	Waste Wood	102	59
Olmstead Cnty. WTE	Rochester	MN	USA	1	42,000	5.5	80	1985	WTE	138	58
Chicago Northwest WTE	Chicago	IL	USA	1	42,000		90	1986	WTE		
SEMASS WTE	Rochester	MA	USA	54	407,500	3.5	59	1986	WTE	120	61
Haverhill RRF	Haverhill	MA	USA	46.9	351,830	5	85	1987	WTE	134	49
Cochrane Sta.	Cochrane	Ont.	CAN	10.5	90,000	3	60	1988	Combined Cycle Cogen	115	55
Grumman	Bethpage	NY	USA	13	105,700	5.4	59	1988	Combined Cycle Cogen	137	78
North Branch Power Sta.	North Branch	WV	USA	80	662,000	7	90	1989	Coal	147	57
Sayreville Cogen Pro.	Sayreville	NJ	USA	100	714,900	3	59	1989	Combined Cycle Cogen	115	56
Bellingham Cogen Pro.	Bellingham	MA	USA	100	714,900	3	59	1989	Combined Cycle Cogen	115	56
Spokane RRF	Spokane	WA	USA	26	153,950	2	47	1989	WTE	102	55
Exeter Energy L.P. Pro.	Sterling	CT	USA	30	196,000	2.9	75	1989	PAC System	114	39
Peel Energy from Waste	Brampton	Ont.	CAN	10	88,750	4.5	68	1990	WTE	130	62
Nipogen Power Plant	Nipogen	Ont.	CAN	15	169,000	3	59	1990	Combined Cycle Cogen	115	56
Linden Cogen Pro.	Linden	NJ	USA	285	1,911,000	2.44	54	1990	Combined Cycle Cogen	108	54
Maalaea Unit 15	Maui	HI	USA	20	158,250	6	95	1990	Combined Cycle	141	46
Norcon Welsh Plant	North East	PA	USA	20	150,000	2.5	55	1990	Combined Cycle Cogen	109	54
Univ of Alaska	Fairbanks	AK	USA	10	46,000	6	82	1991	Combined Cycle Cogen	141	59
Union County RRF	Union	NJ	USA	50	357,000	8	94	1991	WTE	152	58
Saranac Energy	Saranac	NY	USA	80	736,800	5	90	1992	Combined Cycle Cogen	134	44
Onondaga County RRF	Onondaga	NY	USA	50	258,000	3	70	1992	WTE	115	45
Neil Simpson II Sta.	Gillette	WY	USA	80	548,200	6	66	1992	Coal	141	75
Gordonsville Plant	Gordonsville	VA	USA	50	349,150	6	90	1993	C-Cycle (x2 Units)	141	51
Dutchess County RRF Exp.	Poughkeeksie	NY	USA	15	49,660	5	79	1993	WTE	134	55
Samalayuca II Power Sta.	Samalayuca		MEX	210	1,296,900	7	99	1993	Combined Cycle	147	48
Potter Station	Potter	Ont.	CAN	20	181,880	3.8	66	1993	Combined Cycle	124	58

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Facility Name	City	State	Country	Size MW	Steam Flow lbs/hr	Turbine Exhaust Pressure In. Hg	Design Temp. °F	Year	Description	Sat. Steam Temp. °F	Temp. Difference °F
Streeter Generating Sta.	Cedar Falls	IA	USA	40	246,000	3.5	50	1993	Coal - PAC System	120	70
MacArthur RRF	Ronkonkoma	NY	USA	11	40,000	4.8	79	1993	WTE	132	53
North Bay Plant	North Bay	Ont.	CAN	30	245,000	2	53.6	1994	Combined Cycle	102	48.4
Kapuskasing Plant	Kapuskasing	Ont.	CAN	30	245,000	2	53.6	1994	Combined Cycle	102	48.4
Haverhill RRF Exp.	Haverhill	MA	USA	46.9	44,500	5	85	1994	WTE	134	49
Arbor Hills Landfill Gas Fac.	Northville	MI	USA	9	87,309	3	50	1994	Combined Cycle	115	65
Pine Bend Landfill Gas Fac	Eden Prairie	MN	USA	6	58,260	3	50	1994	Combined Cycle	115	65
Pine Creek Power Sta.	Pine Creek	N. Ter.	AUSTRAILIA	10	95,300	3.63	77	1994	Combined Cycle	122	45
Cabo Negro Plant	Punta Arenas		CHILE	6	74,540	4	63	1995	Methanol Plant	126	63
Emeraldas Refinery	Emeraldas		EQUADOR	15	123,215	4.5	87.3	1995	Combined Cycle	130	42.7
Mallard Lake Landfill Gas	Hanover Park	IL	USA	9	101,400	3	49	1996	Combined Cycle	115	66
Riyadh Power Plant 9	Riyadh		SAUDI ARABIA	107	966,750	16.5	122	1996	C-Cycle (x4 Units)	184	62
Barry CHP Project	Barry	S. Wales	UK	100	596,900	3	50	1996	Combined Cycle	115	65
Zorlu Enerji Project	Bursa		TURKEY	10	83,775	3.5	59	1997	Combined Cycle	120	61
Tucuman Power Sta.	El Bracho	Tucuman	ARGENTINA	150	1,150,000	5	99	1997	PAC System	134	35
Dighton Power Project	Dighton	MA	USA	60	442,141	5.5	90	1997	Combined Cycle	139	49
El Dorado Energy	Boulder	NV	USA	150	1,065,429	2.5	67	1998	Combined Cycle	109	42
Tiverton Power Project	Tiverton	RI	USA	80	549,999	5	90	1998	Combined Cycle	134	44
Coryton Energy Project	Corringham		ENGLAND	250	1,637,312	2.5	50	1998	Combined Cycle	109	59
Rumford Power Project	Rumford	ME	USA	80	545,800	5	90	1998	Combined Cycle	134	44
Millmerran Power Project	Toowoomba	Queensland	AUSTRAILIA	420	2,050,000	5.43	88	1999	Coal (x 2 Units)	137	49
Bajio Power Project	Quertetaro	Guananjuaro	MEX	450	1,307,000	3.54	71.4	1999	Combined Cycle	121	49.6
Monterrey Cogen Project	Monterrey		MEX	80	671,970	5.8	102	1999	Combined Cycle Cogen.	140	38
Gelugor Power Station	Penang		MALAYSIA	120	946,600	6.8	89.6	2000	Combined Cycle Cogen.	146	56.4
Front Range Power Project	Fountain	CO	USA	150	1,266,477	3.57	80	2000	Combined Cycle	121	41
Goldendale Energy Project	Goldendale	WA	USA	110	678,000	5	90	2000	C-Cycle PAC System	134	44
Athens Power Station	Athens	NY	USA	120	749,183	5	90	2000	Combined Cycle	134	44
					Average	4				Average	54
					Min	2				Min	35
					Max	16.5				Max	78
HIGH EXHAUST PRESSURE (Temperature Difference >80 °F)											
Beneccia Refinery	Beneccia	CA	USA	NA	48,950	9.5	100	1975		191	91
Beluga Unit 8	Beluga	AK	USA	65	478,400	5.6	35	1979	Combined Cycle	138	103
Univ. of Alberta	Edmonton	Alberta	CAN	25	277,780	9.15	59	1999	Gas Cogen.	158	99

As with wet cooling towers, the ambient air temperature and system design can have an effect on the steam turbine exhaust pressure, which in turn affects the turbine efficiency. Thus, the turbine efficiency can change over time as the air temperature changes. The fans used to mechanically force air through the condenser represent the greatest operational energy requirement for dry cooling systems.

A design measure comparable to the approach value used in wet towers is the difference between the design dry bulb temperature and the temperature of saturated steam at the design turbine exhaust pressure. In general, a larger, more costly dry cooling system will produce a smaller temperature difference across the condenser and, therefore, a lower turbine exhaust pressure. Three facilities in Table 4-1 had high temperature differences (>80 °F), which represent less efficient systems. Two of these facilities are from very cold climates where high temperature differences across the condenser are acceptable and one was for an industrial process (petroleum refining). The range in the temperature difference values for the remaining facilities was 35 to 78 °F. The average was 54 °F.

Steam turbines are designed to operate within certain exhaust pressure ranges. In general, steam turbines that are designed to operate at the exhaust steam pressure ranges typical of wet cooling systems, which generally operate at lower exhaust pressures (e.g., <5 in Hg), may be damaged if the exhaust pressure exceeds a certain value. New steam turbine facilities that are designed to condense steam with dry cooling systems can be equipped with steam turbines that are designed to be safely operated at higher exhaust pressures. EPA has assumed that the difference in costs for turbines that operate over different exhaust pressure ranges are insignificant compared to the total compliance cost and, therefore, no net compliance costs are estimated for the steam turbines.

The data in Table 4-1 shows that turbine exhaust pressures at the highest design dry bulb temperatures in the U.S. (which were around 100 °F) ranged from 5.0 to 9.5 inches Hg. The highest value of 9.5 inches Hg was for a refinery power system in California which, based on the steam rate, was comparable to other relatively small systems generating several megawatts and apparently did not warrant the use of an efficient cooling system. The other data show turbine exhaust pressures of around 6 to 7 inches Hg at dry bulb temperatures of around 100 °F. Maximum exhaust pressures in the range of 8 to12 inches Hg may be expected in hotter regions of the U.S.(Hensley 1985). An air cooled condenser analysis (Weeks 2000) reports that for a combined cycle plant built in Boulder City, Nevada, the maximum ambient temperature used for the maximum off-design specification was 108 °F with a corresponding turbine exhaust pressure of 7.8 inches Hg. Note that the equation used by EPA to generate the turbine exhaust pressure values in the energy penalty analysis produced an estimated exhaust pressure of 8.02 inches Hg at a dry bulb temperature of 108 °F. For wet towers, the typical turbine exhaust pressure operating range is1.5 to 3.5 inches Hg(Woodruff 1998).

For coal-fired plants, the largest operating plant in the United States with dry cooling is the Wyodak Station in Gillette, WY with a total cooling capacity of 330 MW (1.88 million lb/hr of steam). EPA notes that this is significantly smaller than 10 of the projected coal-fired power plants within the scope of the rule and slightly smaller than 25 of the combined cycle plants. The design temperature of the dry system at this plant (which directly affects the size of the dry cooling system) is below average for summer conditions throughout the United States (the Wyodak Station has a design temperature of 66 deg F, whereas recent combined-cycle systems in Rhode Island, Massachusetts, and New York have design targets above 90 deg F). EPA notes that the reported driving force behind the Wyodak Station's decision to utilize dry cooling was the fact that the plant designers wished to locate the plant immediately adjacent to a remote coal-mine mouth.

A demonstrated dry cooling system frequently recognized as the largest in the U.S. is the Linden Cogeneration Plant, in NJ. This cogeneration unit has a comparable cooling capacity to that of a small-sized coal-fired facility (such as the Wyodak Station described above). The cogeneration plant has a total steam flow which requires condensing of

1.91 million lb/hr, which just slightly exceeds the steam flow of the Wyodak station (1.88 million lb/hr). Despite the fact that the Linden plant is designed for a total generating capacity of 640 MW, only 285 MW requires steam condensing. This is because cogeneration units are designed to deliver steam to adjacent manufacturing plants for their use in processes. Therefore, the cogeneration plant has been designed such that only a portion of its steam generation requires cooling, and, for the purposes of evaluating the feasibility of dry cooling, EPA considers this a 285 MW dry cooling facility. EPA notes that the decision for this plant to adopt dry cooling over wet cooling related primarily to a highway safety issue and the visible plume of steam.

Several new combined-cycle projects with dry cooling are either planned or under-construction in the Northeastern US. EPA is aware of eight new dry cooling projects at combined cycle plants in this region that have 350 MW or greater of total plant capacity. The largest of these projects is the permitted Sithe Mystic Station in Massachusetts, which will be a 1500 MW combined-cycle plant. Because the project will utilize a combined-cycle, approximately 500 MW of steam power would require cooling. This will be the largest dry cooling system in the US when complete. However, the system size does not approach the projected cooling requirements for a majority of the coal-fired plants within the scope of this rule.

4.2 IMPACTS OF DRY COOLING

In establishing best technology available for minimizing adverse environmental impact for the final rule, EPA considered an alternative based on a zero-intake flow (or nearly zero, extremely low flow) requirement commensurate with levels achievable through the use of dry cooling systems. In evaluating dry cooling-based regulatory alternatives, EPA analyzed a zero or nearly zero intake flow requirement based on the use of dry cooling systems as the primary regulatory requirement in all waters of the U.S. The Agency also considered subcategorization strategies for the new facility regulation based on size and types of new facilities and location within regions of the country, since these factors may affect the viability of dry cooling technologies. In its evaluation, the Agency considered factors including the demonstration of existing or planned dry cooling systems, the reductions in cooling water intake flow, the environmental and energy impacts, and the associated costs of dry cooling systems.

4.2.1 Cooling Water Reduction

A dry cooling system will achieve an average reduction in cooling water intake flow greater than 99 percent over a once-through system. In comparison, the average flow reduction of a closed-cycle wet cooling system for an estuarine/tidal source is approximately 92 percent, and is 95 percent for a freshwater source. Dry cooling systems therefore achieve an incremental flow reduction from closed-cycle wet cooling to dry cooling of 4 to 7 percent.

4.2.2 Environmental and Energy Impacts

Dry cooling has the benefit of eliminating visual plumes, fog, mineral drift, and water treatment and disposal issues associated with wet cooling towers. The disadvantages of dry cooling include an increase in noise generation and decrease in efficiency of electricity generation which lead to an increase in air emissions as compared to wet cooling systems.

EPA notes that dry cooling systems in all climates are less efficient at removing heat than comparable wet-cooling systems. The practical limitations of the dry cooling system, as limited by the dry bulb temperature, which is always equal to or greater than the wet bulb temperature met by wet cooling systems, prevent its performance from exceeding that of wet cooling. Moreover, increased parasitic fan loads for dry cooling systems will ensure that the technology will not operate as efficiently as a comparable wet cooling system.

Therefore, EPA assessed the negative environmental impacts caused by this loss of efficiency. For combined-cycle plants the mean annual energy penalty (averaged across climates) is 2.1 percent for dry cooling compared to once-through systems, and 1.7 percent for wet cooling compared to once-through systems. For coal-fired plants, the mean annual energy penalty (averaged across climates) is 8.6 percent for dry cooling compared to once-through systems, and 6.9 percent for wet cooling compared to once-through systems. However, for many specific cases, the energy penalty may be dramatically higher for dry cooling due to climatic conditions of the cooling towers. For example, the peak summer shortfalls during hot periods can be debilitating in certain climates due to the energy penalty reaching up to 12.3 percent. See Chapter 3 of this document for further discussion of energy penalties.

EPA projects that a dry cooling based regulatory alternative would result in 1900 MW of lost energy. This is the equivalent electricity generation of two very large (or three large) power plants that would need to be constructed to overcome the energy losses of the dry cooling alternative. The air emissions increases as a result of this replacement capacity, if they were to come from increased generation across the US market, would be equivalent to those of three new 800MW coal-fired power plants. Alternatively, if the replacement capacity comes from new capacity exclusively, it would be from dry cooling equipped plants with the associated elevated capital and annual costs and land area requirements. Therefore, EPA considers the issue of inefficiency of dry cooling, and EPA's subsequent rejection of the dry cooling alternative, to be principal to the concept of energy conservation. Considering that the State of California recently experienced shortages of demand less than the energy penalty of the dry cooling option, the imposition of 1900 MW of mean annual energy penalty capacity loss on planned new power plants does not support the Administration's Energy Plan and associated Executive Orders.

The efficiency of the electricity generation process is directly affected by the cooling system to be installed. The vast majority of projected new plants (i.e., 90 percent) would install closed-cycle recirculating cooling towers regardless of the requirements of this rule. Therefore, EPA's technology-based performance requirements for the final rule based on recirculating closed-cycle cooling would have little impact on the majority of new plants. The flow reduction requirements of the rule are projected to impose changes in cooling system designs on only nine new plants. The comparable effect on the efficiency of these plants will be small on a facility level and national basis.

In contrast, a regulatory alternative based on dry cooling is projected to impose cooling system design changes on each of the 83 power plants within the scope of the final rule. Therefore, each of the 14 projected coal-fired plants would experience mean annual energy penalties ranging from 6.9 to 8.6 percent. The typical steam electric generator (such as modern coal-fired plants) would, at peak operation, operate at less than 40 percent efficiency. The energy penalty of nearly 9 percent is very significant when compared to the system-wide energy efficiency of this type of power plant. Additionally, each of the 69 projected new combined-cycle plants would experience mean annual energy penalties ranging from 1.7 to 2.1 percent. With new design efficiencies of 60 percent, at peak operating efficiency, a 2.1 percent energy penalty is less striking than in the coal-fired cases. However, the cumulative effect for all 69 power plants is substantial.

4.2.3 Costs of Dry Cooling

The final rule analysis, which includes the contribution of the energy penalty to the recurring annual costs, projects that the total annualized cost for the dry cooling alternative is $490 million (in 2000 dollars). EPA notes that the vast majority of costs associated with this option are incurred at the 83 power plants, and not at the 38 manufacturers subject to this rule. Because dry cooling is not a feasible option for all manufacturing facilities, EPA only applied costs of recirculating wet cooling towers to these types of facilities. The present value of total compliance costs for drying cooling are projected to be $6 billion.

A comparison of capital costs between equally sized combined-cycle plants for wet and dry cooling tower systems reveals that the dry cooling plant's capital costs would exceed those of the wet cooling tower plant by 3.3 fold. The installed wet cooling tower capital cost is approximately $10 million, while the dry cooling installation would cost approximately $33 million. For a typical, modern 700-MW combined-cycle power plant, the erected capital costs for a wet cooling tower represent approximately 2 percent of the total capital costs of the power plant construction project compared to 6.5 percent for dry cooling towers.

EPA also evaluated a comparison of the operation and maintenance costs associated with these two types of cooling systems for an equally sized combined-cycle model plant. The operation and maintenance costs of the wet cooling tower (without including the effects of energy penalties) would be $1.8 million per year, while the dry cooling system would cost $7.4 million per year. Without incorporating energy penalties, the ratio of operation and maintenance costs of dry cooling to wet cooling for a typical 700-MW combined-cycle power plant would be greater than 4 to 1. After factoring in the recurring costs of energy penalties for the two systems, the recurring annual costs increase to $2.3 million for the wet tower plant and $10.4 million for the dry cooling plant. This corresponds to a dry to wet ratio also greater than 4 to 1. The total annualized costs for this model facility are estimated at $3.1 for the wet cooling tower system and $13.1 for the dry cooling system (a ratio of 4.2 to 1). Note that these are comparative cost estimates for a hypothetical facility and do not represent actual compliance costs of the rule.

4.2.4 Methodology for Dry Cooling Cost Estimates

EPA estimated the capital and O&M costs using relative cost factors for various types of wet towers and air cooled condensers, using the cost of a comparable wet tower constructed of Douglas Fir as the basis. Chapter 2 provides the capital and operating cost factors that were used by EPA. These cost factors were developed by industry experts who are in the business of manufacturing, selling and installing cooling towers, including air cooled systems, for power plants and other applications. For air cooled condensers (constructed of steel), a range of cost factors is given in Table 4-3. EPA based the capital and O&M costs on these factors with some modifications. To be conservative, EPA chose the highest value within each range as the basis. The factors chosen are 325 percent and 225 percent (of the cost of a mechanical wet tower) for capital cost (for a tower with a delta of 10 ºF) and O&M cost, respectively. EPA applied a multiplier of roughly 1.7 to the dry tower capital cost estimates for a delta of 10 ºF to yield capital cost estimates for a dry tower with a delta of 5 ºF. EPA applied these factors to the capital costs derived for the basic steel mechanical draft wet cooling towers to yield the capital cost estimates for dry towers presented in Table 4- 2.

Note that the source document for these factors states that the factors represent comparable cooling systems for plants with the same generated electric power and the same turbine exhaust pressure. Since the cost factors generate equivalent dry cooling systems, the tower costs can still be referenced to the corresponding equivalent cooling water flow rate of the mechanical wet tower used as the cost basis. Since the final §316(b) New Facility Rule focuses primarily on water use, the use of the cooling flow or the "equivalent" was considered as the best way to compare costs. The costing methodology uses an equivalent cooling water flow rate as the independent input variable for costing dry towers.

Table 4-2: Estimated Capital Costs of Dry Cooling Towers with Delta of 5 °F and 10 °F (1999 Dollars)

Flow (gpm)	Delta 5 ºF	Delta 10 ºF
2000	$790,000	$450,000
4000	$1,580,000	$949,000
7000	$2,766,000	$1,658,000
9000	$3,556,000	$2,132,000
11,000	$4,345,000	$2,607,000
13,000	$5,135,000	$3,081,000
15,000	$5,925,000	$3,556,000
17,000	$6,715,000	$4,027,000
18,000	$7,108,000	$4,264,000
22,000	$8,515,000	$5,038,000
25,000	$9,675,000	$5,727,000
28,000	$10,836,000	$6,412,000
29,000	$11,222,000	$6,643,000
31,000	$11,996,000	$7,101,000
34,000	$13,156,000	$7,787,000
36,000	$13,933,000	$8,245,000
45,000	$17,059,000	$9,952,000
47,000	$17,817,000	$10,394,000
56,000	$21,229,000	$12,383,000
63,000	$23,881,000	$13,933,000
67,000	$25,399,000	$14,817,000
73,000	$27,674,000	$16,143,000
79,000	$29,325,000	$16,845,000
94,000	$34,892,000	$20,043,000
102,000	$37,859,000	$21,749,000
112,000	$41,574,000	$23,881,000
146,000	$54,194,000	$31,132,000
157,000	$57,034,000	$32,237,000
204,000	$72,498,000	$40,277,000
250,000	$100,800,000	$58,800,000
300,000	$120,000,000	$70,000,000
350,000	$140,400,000	$81,900,000
400,000	$160,800,000	$93,800,000

Using the estimated costs, EPA developed cost equations using a polynomial curve fitting function. Table 3 presents capital cost equations for dry towers with deltas of 5 and 10 degrees.

Table 4-3. Capital Cost Equations of Dry Cooling Towers with Delta of 5 °F and 10 °F

Delta	Capital Cost Equation[1]	Correlation Coefficient
5 °F	$y = -2E\text{-}10x^3 + 0.0002x^2 + 337.56x + 973608$	$R^2 = 0.9989$
10 °F	$y = -8E\text{-}11x^3 + 0.0001x^2 + 189.77x + 800490$	$R^2 = 0.9979$

1) x is for flow in gpm and y is cost in dollars.

For purposes of estimating costs for the dry cooling option (Option 2B) for the final §316(b) New Facility Rule, EPA used the O&M cost curve for air condensers contained in Appendix A of the *Economic and Engineering Analyses of the Proposed §316(b) New Facility Rule* without modification. Thus, EPA overcosted the O&M costs for dry towers for Option 2B for the final §316(b) New Facility Rule. See Section 2.9.1 of this document and the response to comment document (#316bNFR.068.330) for discussion of EPA's revised O&M costs for the final rule.

Validation of Dry Cooling Capital Cost Curves

To validate the dry tower capital cost curves and equations, EPA compared the costs predicted by the equation for dry towers with delta of 10 ºF to actual costs for five dry tower construction projects provided by industry representatives. To make this comparison, EPA first needed to estimate equivalent flows for the dry tower construction project costs. Obviously, as noted above, dry towers do not use cooling water. However, for every power plant of a given capacity there will, dependent on the selected design parameters, be a corresponding equivalent recirculating cooling water flow that would apply if wet cooling towers were installed to condense the same steam load.

EPA used the steam load rate and cooling system efficiency to determine the equivalent flow. Note that the heat rejection rate will be proportional to the plant capacity. EPA estimated the flow required for a wet cooling tower that is functionally equivalent to the dry tower by converting each plant's steam tons/hour into cooling flow in gpm using the following equations:

Steam tons/hr x 2000 lbs/ton x 1000 BTUs/lb steam = BTUs/hr
One ton/hr = 12,000 BTU/hr
BTUs/hr / 12000 = Tons of ice
Tons of Ice x 3 = Flow (gpm) for wet systems

Chart 4-2 presents a comparison of the EPA capital cost estimates for dry towers with delta of 10 ºF (with 25% error bars) to actual dry tower installations. This chart shows that EPA's cost curves produce conservative cost estimates, since the EPA estimates are greater than all of the dry tower project costs based on the calculated equivalent cooling flow rate for the actual projects.

Chart 4-1. Capital Costs of Dry Cooling Towers Versus Flows Of Replaced Wet Cooling Towers (5 &10 Degrees Delta)

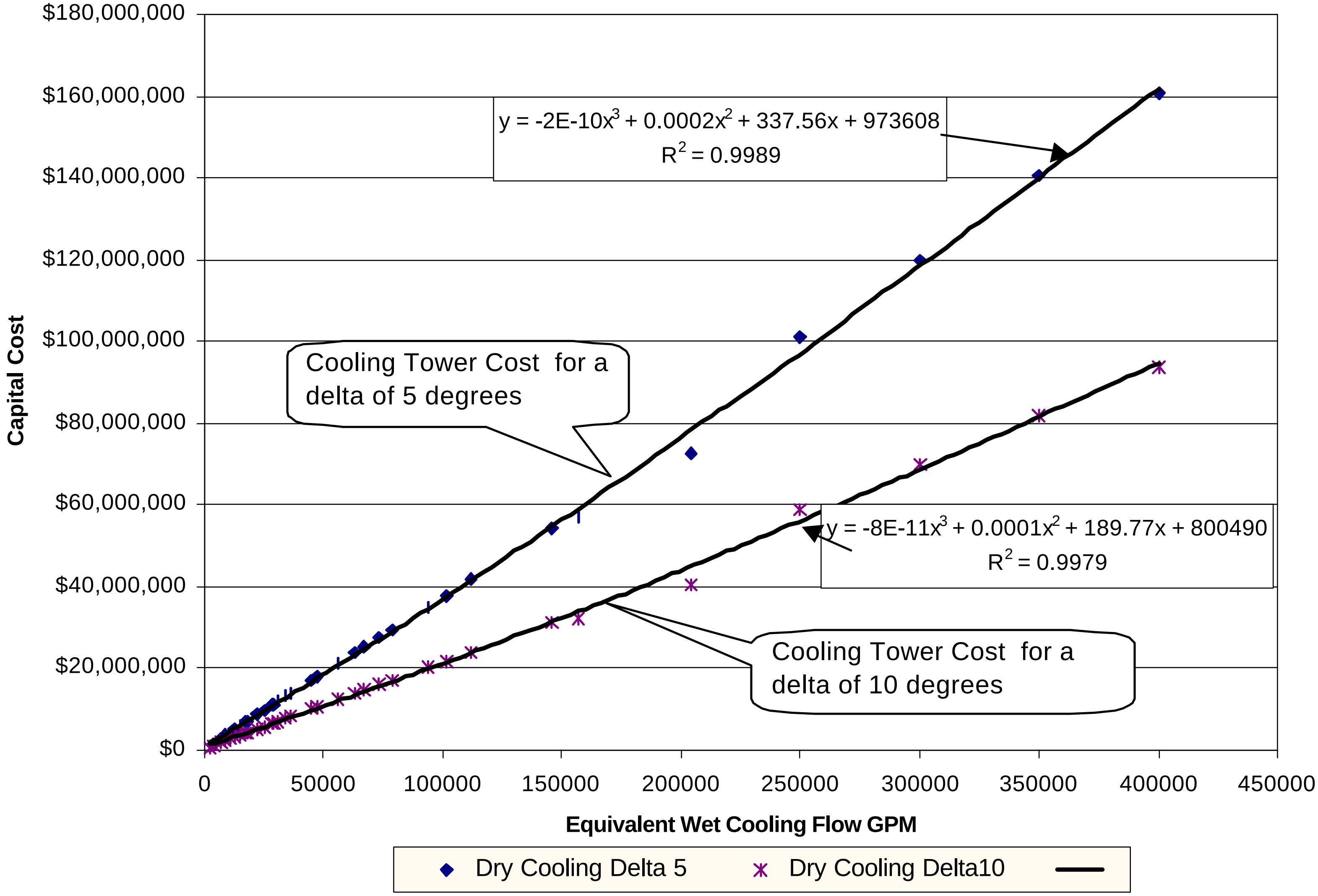
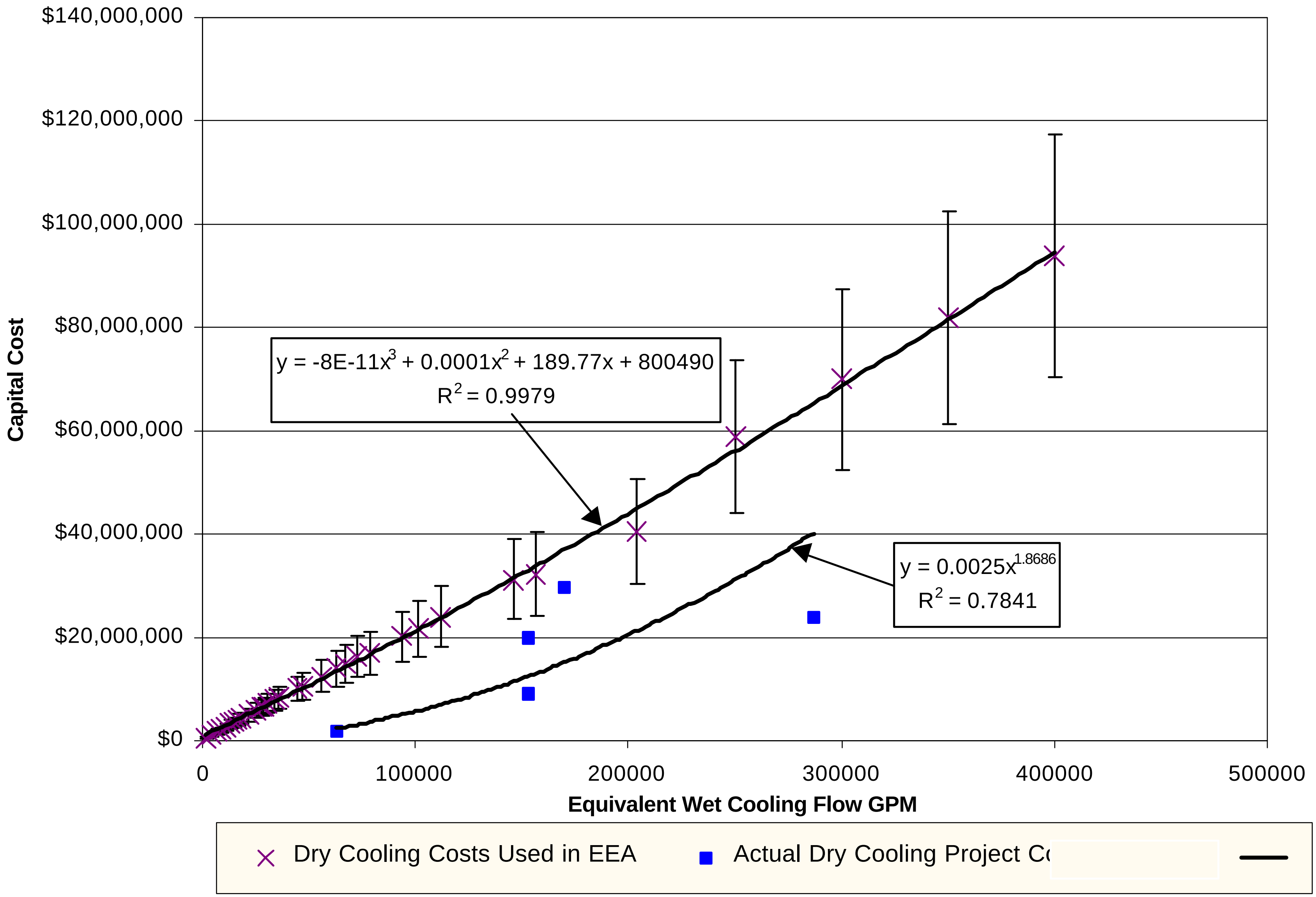


Chart 4-2. Actual Capital Costs of Dry Cooling Tower Projects and Comparable Costs from EPA Cost Curves



4.1.6 Economic Impacts of Dry Cooling

EPA concluded that the costs of dry cooling systems may be significantly prohibitive so as to pose barriers to entry for some new plants. EPA projected that the cost to revenue impacts exceed 10 percent for 12 new power plants and exceed 4 percent for all new plants under a dry cooling-based regulatory alternative. EPA considers this level of cost to revenue impacts to be significant. In comparison, the cost to revenue impacts of the final rule, which is based in part on flow reduction commensurate with that achieved using recirculating closed-cycle wet cooling, do not exceed 3 percent for a single facility, and the vast majority of the impacts are below 1 percent. A complete discussion of the cost to revenue impacts and discussion of barrier to entry analysis can be found in the Economic Analysis for the final rule. As such, regional subcategorization options would pose similar barriers to entry for new plants in the Northeastern United States, combined with imposing competitive disadvantages for the subset of facilities complying with more stringent and costly standards than the other regions of the country.

EPA is concerned that the barrier to entry, high costs, and energy penalty of dry cooling systems may remove the incentive for replacing older coal-fired power plants with more efficient and environmentally favorable new combined-cycle facilities. By basing the requirements of the rule on dry cooling, regulated entities faced with the prospects of building new facility power plants that are required to utilize dry cooling would, instead of beginning or continuing with the new facility project, turn to existing power-plants (many of which are significantly aged) and attempt to extend their operating lives further or refurbish them such that the new facility rule would not apply.

EPA notes that there have been recent advances in the efficiency of power plants, specifically combined-cycle plants, that have many environmental advantages. Combined-cycle plants produce significantly less air emissions of NOx, SO_2, and Hg per MWh generated, use less water for condensing of steam than fossil-fueled or nuclear plants (greater than one-half water use reduction per MWh of generation), and are significantly more energy efficient in their generation of electricity than comparable coal-fired plants. The Agency does not wish to create disincentives for the construction of new efficient plants such as these.

4.3 Evaluation of Dry Cooling as BTA

This section presents a summary of EPA's evaluation of the dry cooling technology as a candidate for best technology available to minimize adverse environmental impacts. Based on the information presented in the previous sections, EPA concluded that dry cooling systems do not represent the best technology available for a national requirement and under the subcategorization strategies described above.

First, EPA concluded that dry cooling is not adequately demonstrated for all facilities within the scope of this regulation. As noted previously, the majority of operating or planned dry cooling systems are located either in colder or arid climates where the average dry bulb temperatures of ambient air is amenable to dry cooling. As demonstrated in Chapter 3, the comparative energy penalty of a dry cooling plant in a hot environment at peak summer conditions can exceed 12 percent at a facility, thereby making dry cooling extremely unfavorable in many areas of the U.S. for some types of power plant types.

EPA's record demonstrates that of the demonstrated, permitted, or planned power plants in the Northeastern United States with dry cooling, the size and capacity of these dry cooling systems is considerably smaller than that necessary to condense the steam load for even below average sized coal-fired power plants projected within the scope of this rule.

Dry cooling technology has a detrimental effect on electricity production by reducing energy efficiency of steam turbines, especially in warmer climates The reduced energy efficiency of the dry cooling system will have the effect of increasing air emissions from power plants.

Lastly, EPA concluded that the costs of dry cooling systems may be significantly prohibitive so as to pose barriers to entry for some new plants that may discourage the construction of new, more energy efficient plants.

In addition to the technical feasibility and cost impacts of dry cooling, EPA also evaluated the expected benefits that would be achieved by dry cooling. EPA notes that the two-track option based on reducing intake flow to a level commensurate with wet cooling towers reduces intake flows by 92 to 95 percent over a once-through system. Dry cooling would only reduce intake flow by an additional 4 to 7 percent. Additionally, the selected option requires velocity and design and construction technology-based performance requirements for the remaining intake flow. These performance requirements are expected to further decrease the negative environmental impacts of the cooling water intake flow, thereby reducing impingement and entrainment of organisms to dramatically low levels. See Chapter 5 for discussion of design and construction technologies to reduce impingement and entrainment.

In summary, EPA concluded that dry cooling is not technically or economically feasible for all facilities subject to this rule, would increase air emissions due to the energy penalty, has a cost more than three times that of the selected regulatory option, and would not significantly reduce impingement and entrainment beyond the regulatory approach selected by EPA to offset these drawbacks. For these reasons, EPA concluded that dry cooling does not represent the "best technology available" for minimizing adverse environmental impact.

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